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ALBA Cooling System Upgrade: Thermal Hydraulic Numerical Simulations

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Abstract -For the period from June 2013 to October 2014 the ALBA Synchrotron has run a transversal project to accomplish an upgrade of the ALBA Cooling System, in order to increase its reliability (more protection against point failure), stability (more robustness in front of load variations) and adequacy in fail mode (maximum service readiness in fail mode). All the activities have been grouped in five work packages: Fluid Dynamics and Thermal Control, Consumption Side Upgrade, Production Side Upgrade, Remote Supervision System, and Cabling and Control System.

This paper deals with the work package “Fluid Dynamics and Thermal Control”. The goal is to advance on the understanding of the cooling system, based on numerical simulations, in order to take some engineering decisions. Two simulations have been performed: (i) Thermo-fluid dynamic simulations of the ALBA Cooling System; and (ii) Numerical evaluation of a Borda mouthpiece mounted in the accumulator tank of the ALBA Cooling System. For this action, ALBA project team has established a deep collaboration with the Centre for Industrial Diagnostics and Fluid Dynamics (CDIF) of the UPC.

For the first simulation a numerical 1D model has been developed by means of the FLOWMASTER software. The model comprises all the elements of the entire network like pumps, pipes, valves, bends, accumulators, junctions, heaters/coolers and other minor components. Moreover, the system automatic regulation mechanisms for temperatures and pressures have also been implemented. The numerical predictions have been compared with experimental data. A good agreement has been found with deviations of the main variables below 10%. For the second simulation; in order to de-aerate the cooling system, the mounting of a Borda mouthpiece at the upper vertical discharge tube of the accumulator tank has been proposed to create a stagnation region where air is expected to accumulate. Then, the air is removed through automatic air release valves. These effects have been evaluated by means of a two-phase flow simulation, based on ANSYS-CFX software.

***Keywords*:** Cooling, Hydraulic, Flowmaster, Ansys-CFX, CFD.

1. Background

After five year in operation the ALBA Synchrotron Light Source (Web-1) implemented the first upgrade process on its Cooling System. The main objective of the project was to accomplish an upgrade of the hydraulic plant and its control system up to an internally defined reasonable level, in order to significantly increase its reliability (more protection against single point failure), stability (more robustness in front of load variations and/or external perturbations) and adequacy in fail mode (maximum service readiness in fail mode).

To face this objective a Project Manager structure was defined (see Figure 1) and all the activities were grouped in the following five work packages (WP) (Quispe, 2014b):

- Fluid Dynamics and Thermal Control (WP1). Goal: To better understand the cooling system from the fluid dynamics and thermal control point of view. Construction of a model used to define the system settings that provide the right behavior with maximum stability.
- Hardware Upgrade (consumption side) (WP2). Goal: To improve (upgrade) instrumentation at the plant. Addition of sensors to better monitor the plant (these sensors are the basis for the new remote supervision system). To improve static and dynamic control elements at hardware level (addition of pressure limiters, bypass circuits, valves, etc.).
- Hardware Upgrade (production side) (WP3). Goal: To improve (adding redundancies) PLC elements at the generation sub-plant. Addition of sensors to better monitor the plant and to improve static elements at hardware level (degassing equipment, air vents, filters, etc.).
- Remote Supervision System (WP4). Goal: To develop a new remote supervision system for the accelerator control room.
- Cabling and Control System Installation (WP5). Goal: Complete the electrical and control interface of the instruments to the Control System.

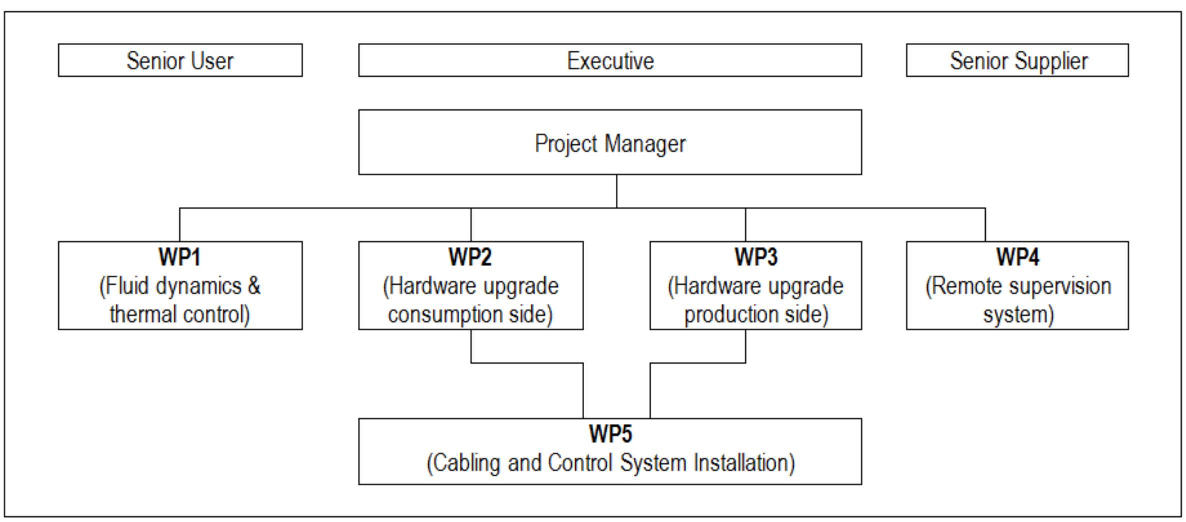

Fig. 1. Organization of work packages for the project “ALBA Cooling System Upgrade”.

The main content of this paper is dedicated to the work package 1 (WP1). For this section two simulations have been performed: (i) Thermo-fluid dynamic simulations of the ALBA Cooling System;

and (ii) Numerical evaluation of a Borda mouthpiece mounted in the accumulator tank of the ALBA Cooling System.

2. ALBA Cooling System Description

The ALBA cooling system is comprised by two main parts: the production and consumption sides. For the refrigeration four groups of pumps feed the rings Experimental Area (EA), Service Area (SA), Storage Ring (SR) and Booster (BO) (see Figure 2). Both the Storage and the Service Area rings operate with a couple of twin-pumps mounted in parallel and the rest with a single pump. The deionized water is heated thorough all the rings and it is collected in a common return line. Another pump (P11) takes the heated water from the return and feeds a couple of heat exchangers that cool it. The cooled water is brought to a large volume accumulator from which a suction line takes water again to the rings' pumps. In order to regulate the water temperature, a series of controlled mixing valves permit to combine the cooled water with the heated water prior to being pumped to the rings. Moreover, a pressure maintenance system with a compressor is mounted at the exit line of the heat exchangers before the accumulator. Finally, a pipe line connecting the accumulator with the common return line permits to compensate the lack/excess of flow to the cooling loop when the total flow rate changes in the rings' loops.

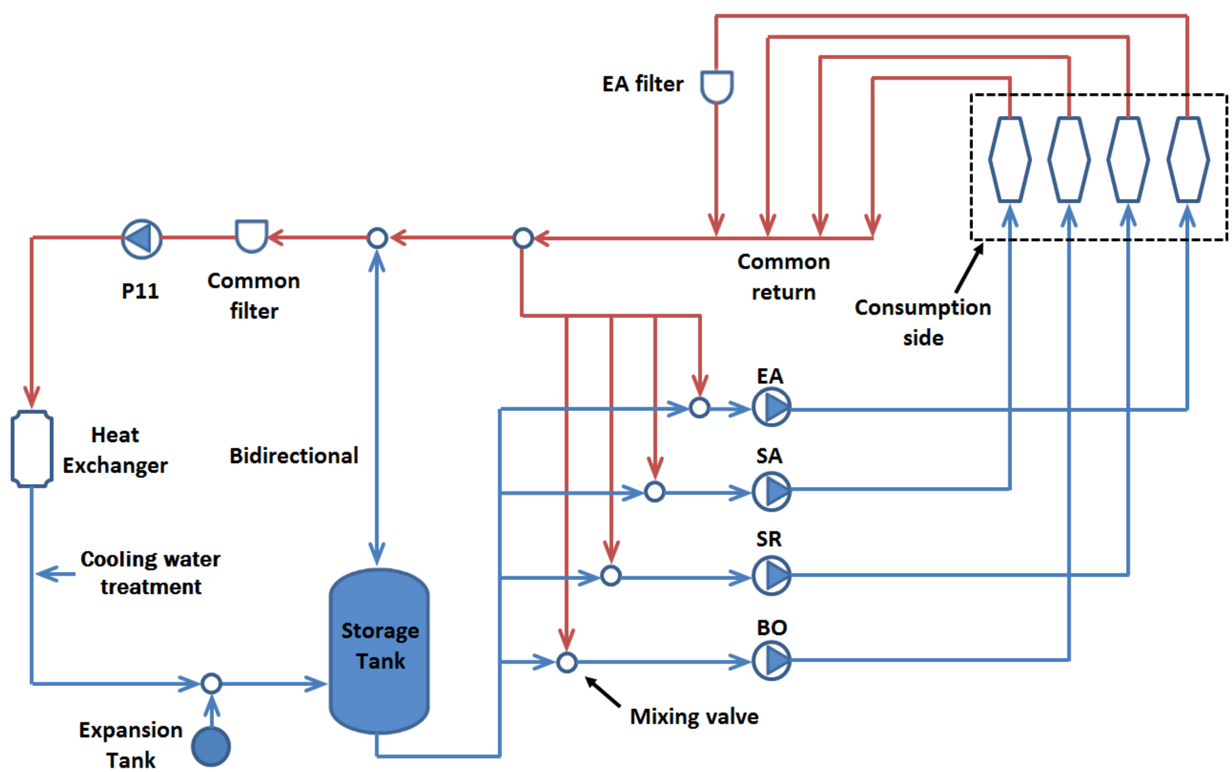

Fig. 2. ALBA Cooling System simplified scheme.

3. Thermo – fluid dynamic simulations

With the aim of simulating the steady state and transient behaviour of the ALBA cooling piping system under typical operating conditions, the Flowmaster® software (Flowmaster Group BV) has been used which provides a 1D modelling solution for the thermo-fluid system properties.

3. 1. General model description

The model has been built up from the available components in Flowmaster software (Figure 3). The properties of each component have been selected based on the information provided by the corresponding manufacturer in the form of construction planes, technical documentation and so on. The lack of reliable information has been overcome with visual inspections and measurements in-situ.

As an example of the level of detail achieved, the pumping subsystems that deliver high pressure water to the rings have been modelled as shown on the left of Figure 4 for the particular case of the Service Area pumps P10. The common elements are the pipes, the junctions, the bends, the transitions and the valves. Other components that are no available in Flowmaster database, like the rubber joints, have been modelled with general purpose components.

However, such level of detail has not been achieved on the rings for the current study due to their dense and complex structure. In this case, it has been decided to simplify each of them with a heat exchanger and a globe valve as shown on the right of Figure 4.

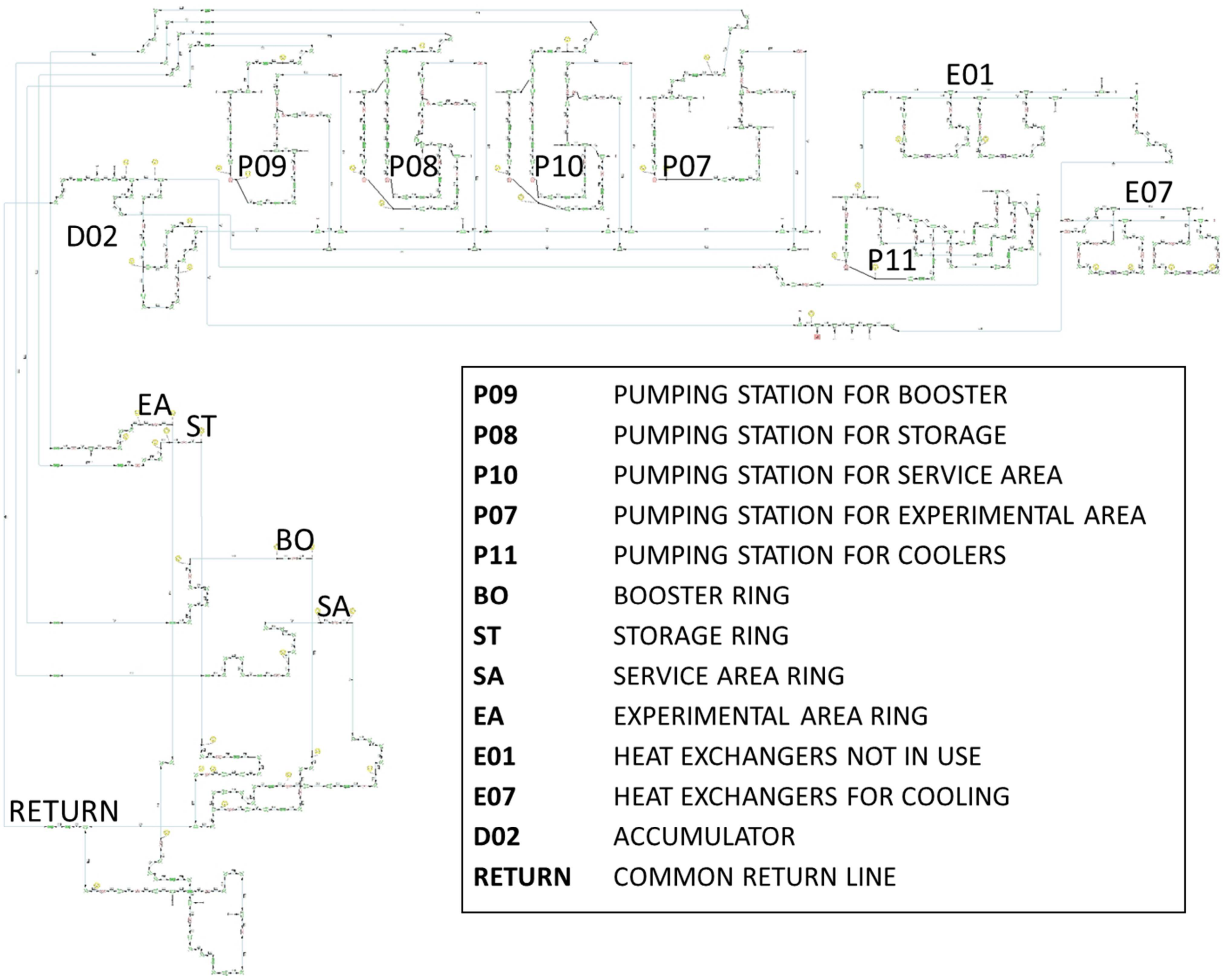

Fig. 3. Schematic of the entire network with description of the subnetworks used in the model.

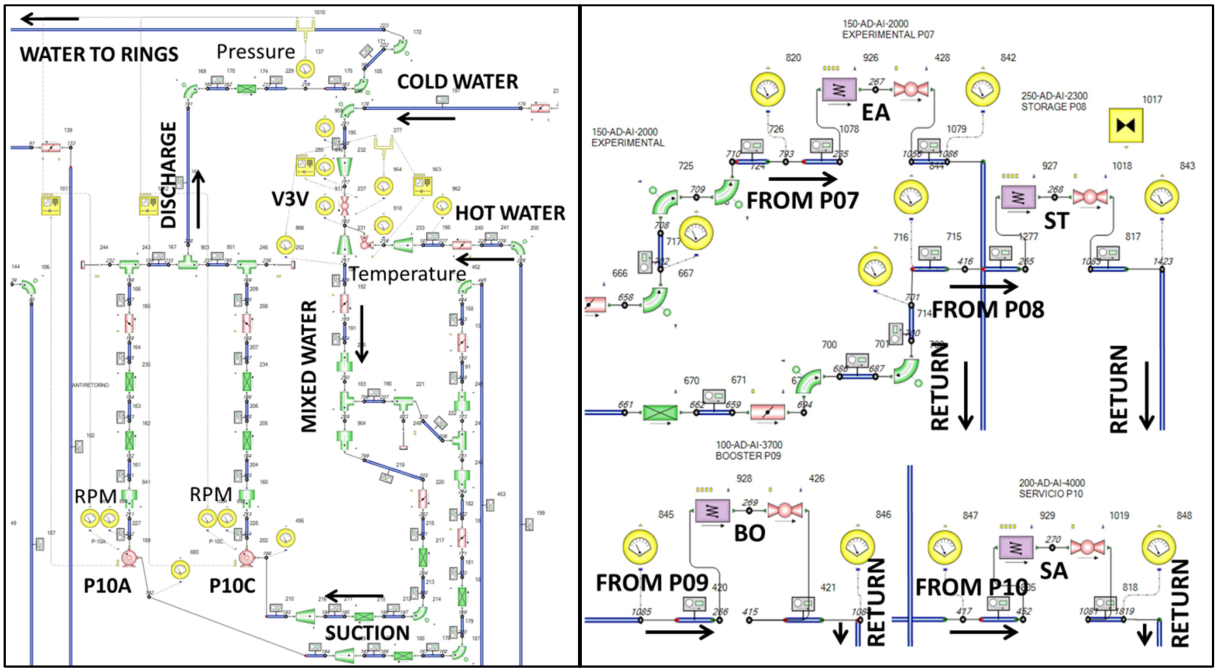

Fig. 4. Schematic of the subnetwork for Service Area pumping station (left) and of the simplified subnetworks for the rings (right).

The cooling system is regulated by means of the simultaneous control of:

1. The flow pressures that feed each ring,
2. The flow temperatures that feed each ring,
3. And the common return flow pressure that is collected from all the rings.

In the model these variables are respectively controlled with:

1. The pump rotating speeds,
2. The opening ratios of the mixing valves (see V3V on the left of Figure 4),
3. And a pressure source element.

3. 2. Accuracy of the model

In order to certify the accuracy of the modelled network the simulated results for various operating conditions were compared with on-site measured data. In particular, four thermo-hydraulic settings (A, B, C and D) (Quispe, 2014a) were considered which comprise, for each ring, the inlet and outlet temperatures (Ti, To), the flow rates (Q), the delivery pressure (P), the heat duty (N), the percentage of cold water coming from the reservoir D02 at the mixing valve (%V3V) and the pump rotating speeds (n). From the obtained results, the goodness of the model is confirmed by the fact that the maximum deviations around the 6 % for the hydraulic variables and around the 10% for thermal variables. An example of some simulated quantities and their percent error relative to the measured values are indicated in Table 1 for thermo-hydraulic setting D.

Table 1: Simulated flow rates (Q), delivery pressures (P) and delivery temperatures (Ti), and their percent error relative to the measured values for thermo-hydraulic setting D.

RING	*Q [m3/h]*	*% error Q*	*P [bar]*	*% error P*	*Ti [°C]*	*% error Ti*
Booster	*28.6*	*0*	*10.2*	*0*	*23.1*	*1*
Storage	*271.1*	*0*	*10.2*	*0*	*23.4*	*-1*
Service Area	*200.1*	*0*	*10.1*	*1*	*23.2*	*0*
Experimental	*16.2*	*0*	*7.1*	*-5*	*22.0*	*5*

3. 3. General system stability

The operation conditions of the cooling system are determined by the heat exchange that takes places across the rings' instrumentation and the particular piping configuration with a common return as outlined in Figure 5. First of all it is assumed that the system cooler heat exchangers have unlimited capacity to provide a constant and continuous outlet temperature of 21°C that fills the accumulator. Secondly, the water temperature requirement at the inlet of the rings is of 23°C. Given the fact that the exit flow rates and temperatures at each ring are variable, it is necessary to continuously adjust the water mixing process by the use of three way mixing valves. These valves mix hot water from the common return and cold water from the accumulator.

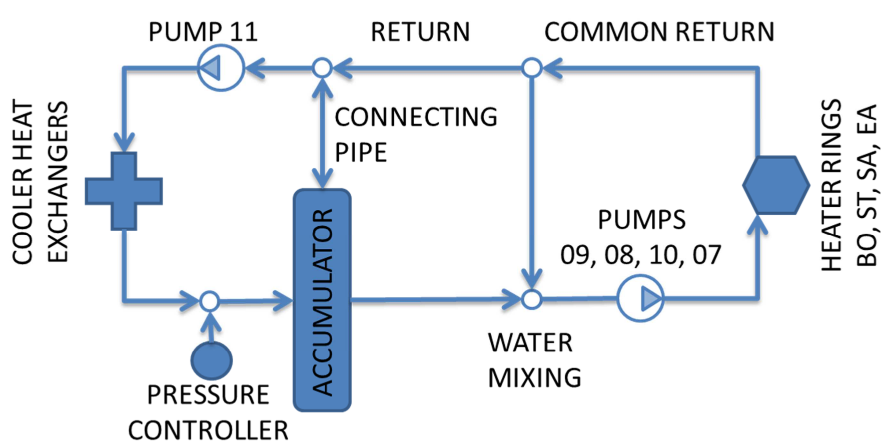

Fig. 5. Outline of the cooling system.

Therefore, the general system behavior can be studied by monitoring the evolution of thermo-fluid properties as a function of changes in the mixing valve positions. With this aim, a series of parametric tests have been carried out changing the valve opening ratios from 0 to 1 in steps of 0.1. In particular, a ratio of 0 indicates that the valve is completely closed to the accumulator and a ratio of 1 indicates that it is completely opened to the accumulator. The obtained mixed flow temperature and flow rate at connecting pipe as a function of valve position are plotted in Figure 6 (left and right respectively).

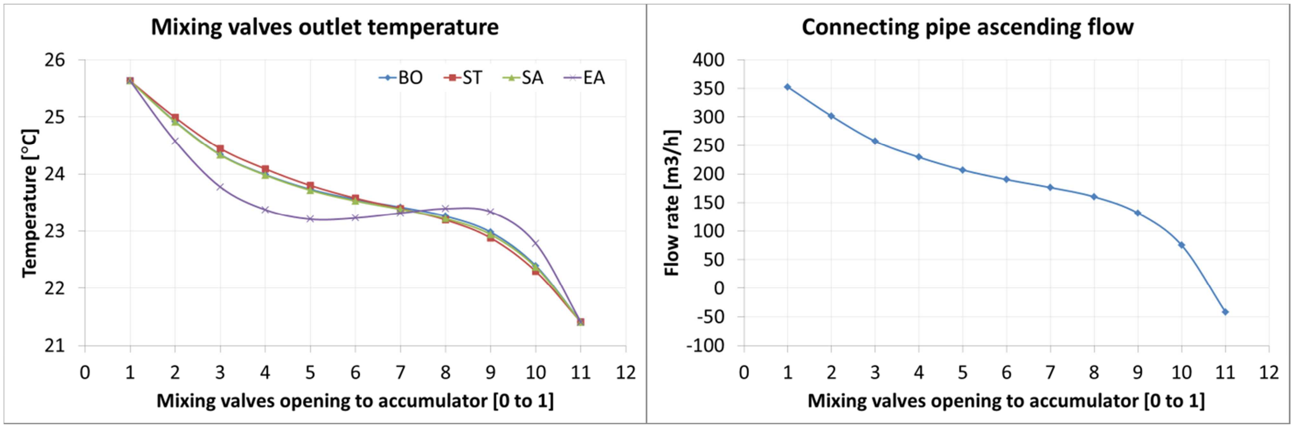

Fig. 6. Outlet temperature of mixed flow at valves and flow rate at connecting pipe as a function of simultaneous valve position changes.

The first observation regarding the behavior of the outlet temperature is the different evolution observed for the Experimental Area subsystem. The temperature set-point is achieved at positions around 0.8 for the Booster, Storage and Service Area and around 0.9 for the Experimental Area. The second important point is the fact that at these operation conditions the flow rate on the connecting pipe is low and close to zero condition. From the system outline in Figure 5 it is obvious that, in the case that this flow reverses and becomes negative, the thermal control is completely lost which can induce general system instability. Consequently, operation close to this condition is not recommended and it must be controlled.

3. 4. Mean flow velocity distributions

To avoid air problems in pipelines it is widely accepted that minimum flow velocities are required above 0.5 m/s (British Standards Institution, 2000). Thus, the simulation of the entire piping system has permitted to identify the locations with lowest velocities. As an example, the results corresponding to a detailed network of the Experimental Area pipeline are given as follows.

In particular, this pipeline consists of two concentric rings that can feed 20 subsystems. Nowadays, 7 of them correspond to Beam Lines (BL), 2 of them are by-passed (BP) and the rest (11) are closed connections not in use (C). Originally, the inlet flow is equally distributed to the left branch (clockwise) and to the right branch (counter clockwise) through a T-junction by opening the two exit valves. Then, for the outlet flow, the directions are reversed and the two flows are converged to the main outlet pipe. This original flow distribution is indicated as 180° circulation on the left of Figure 7. Nevertheless, this configuration tends to reduce the main flow velocities as the two flows approach and ideally a zero velocity point must be achieved. Therefore, there is a high risk of air accumulation in that zone.

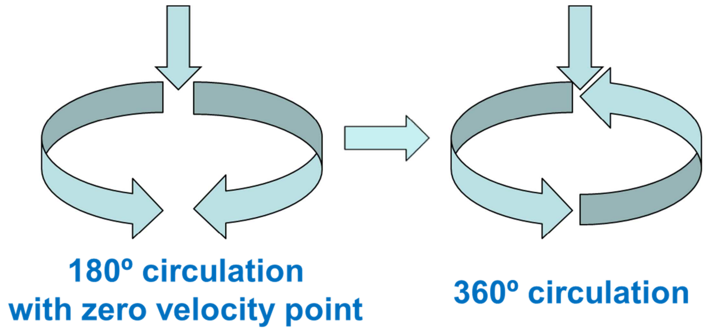

Fig. 7. Schematic of the 180° and 360° flow configurations simulated at Experimental Area rings.

In order to improve the flow velocity distribution along the rings, it was considered to force a 360° circulation as indicated on the right of Figure 7 by closing the one valve of the T-junction exit branches. Doing so, the main flow was forced to circulate only in counter clockwise direction at the inlet ring and in the opposite direction at the outlet ring.

The comparison of mean flow velocities in all the pipes between any two possible connections is given in Table 2. For the 180° circulation all the figures are equal or below 0.5 m/s. Meanwhile, for the 360° configuration, only 8 pipes out of 19 present values below 0.5 m/s. Consequently, the improvement with the new flow configuration is significant.

Table 2: Mean flow velocities in the pipes for 180° and 360° flow configurations simulated at Experimental Area inlet ring.

	Valve-BP1	**BP1-BL29**	**BL29-C1**	**C2-C3**	**C3-BL24**	**BL24-BL22**	**BL22-C4**	**C4-C5**	**C5-C6**	**C6-BL13**	**BL13-BL11**	**BL11-BL09**	**BL09-C7**	**C7-BL04**	**BL04-C8**	**C8-C9**	**C9-C10**	**C10-BP2**	**BP2-Valve**
$v_{180°}$ [m/s]	0,33	0,28	0,22	0,22	0,22	0,19	0,14	0,14	0,14	0,14	0,06	0,01	0,08	0,08	0,24	0,24	0,24	0,24	0,50
$v_{360°}$ [m/s]	0,82	0,77	0,71	0,71	0,71	0,68	0,63	0,63	0,63	0,63	0,55	0,48	0,41	0,41	0,25	0,25	0,25	0,25	0,00

3. 5. Common return influence on system regulation

The fact that the flow through the rings is not fully independent due to the parallel piping configuration with a common return is the root cause of an interdependent behavior that needs to be understood. With this objective, the transient behavior of the overall cooling system under local operation changes has been evaluated. As an example, the effects of a progressive flow reduction in the Storage ring from 271 to 133 m3/h taking place in about 7 seconds are presented in figures 8 and 9.

On the left Figure 8, it is clearly observed how the ring’s flow reduction is compensated by an increase of the connecting pipe flow rate. The lack of return flow that feeds the cooling subsystem is compensated with an increase of accumulator flow. On the other hand, on the right of the same figure it can be seen that all the suction pressures are affected during the transient. Finally, the suction pressure at the pumping system is increased by an amount of 0.1 bars.

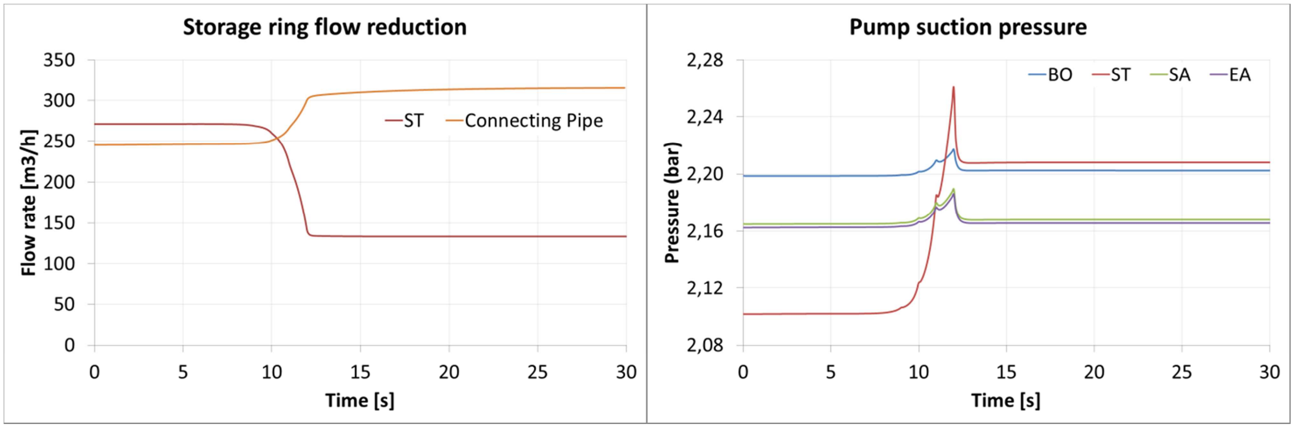

Fig. 8. Flow rate and pump suction pressure evolution with Storage ring flow reduction.

As observed on the left of Figure 9, the flow reduction provokes a progressive closure of all the mixing valves to the cold water from the accumulator and the corresponding opening to the hot water from the return. This is because the temperature of the mixed water cannot be kept constant during the transient as seen on the right of the same figure. In particular, the most affected valve is the one controlling the Storage ring flow. However, the results show that the 30 seconds of time are not enough for the control system to stabilize thermal properties in opposition to the fast balance of some hydraulic parameters. It is clearly observed that at the end of the 30 s no steady state condition has been achieved yet.

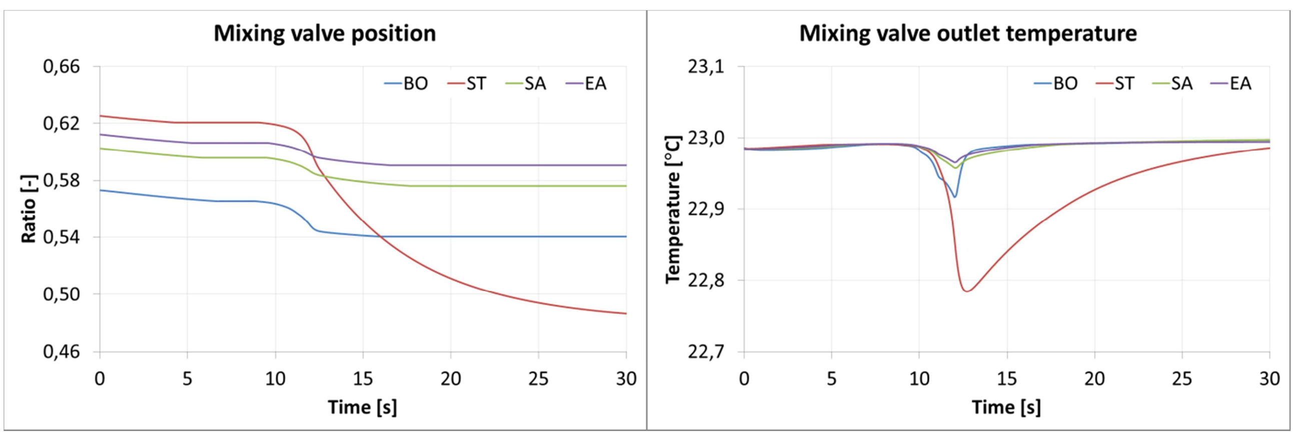

Fig. 9. Mixing valve position and outlet temperature evolution with Storage ring flow reduction.

4. Numerical evaluation of a Borda mouthpiece mounted in the accumulator tank

Serious problems might arise if excessive air is accumulated in a pressurized system. For instance, the risk that air pockets may form in the higher elevated parts of the pipelines is high. As a result, the resistance of the pipe will increase and the flow rate will decrease if spare pump capacity is not available. One way of reducing this risk is to properly de-aerate the piping system (Escarameia et al, 2005). The most common method is the use of Automatic Air Release Valves. The typical locations of these valves are high points along the network where air can be collected by upward bubble displacements due to buoyancy forces. Based on these principles, it was decided to promote air collection in the large accumulator tank by the mounting of a Borda mouthpiece at the top outlet (Dong et al., 1986). The idea behind this modification was to create a stagnation region with low velocities where air would be captured and then removed by a series of release valves. In order to evaluate the feasibility of the idea and to optimize the de-aeration efficiency a series of numerical simulations were carried out with Ansys-CFX software (ANSYS Inc., 2014).

4. 1. Mounting details of Borda mouthpiece and automatic air release valves

The Borda mouthpiece consists of a pipe extension of about 660 mm from the top exit towards the inner part of the accumulator as shown on the drawings in Figure 10. Three holes distributed uniformly around the vertical axis at a radius of about 333 mm were drilled on the upper wall of the accumulator. They connect with the air release valves through vertical tubes. As it can be observed, the tank has a diameter of 2900 mm and there is a lateral flow inlet and two vertical outlets at top and bottom.

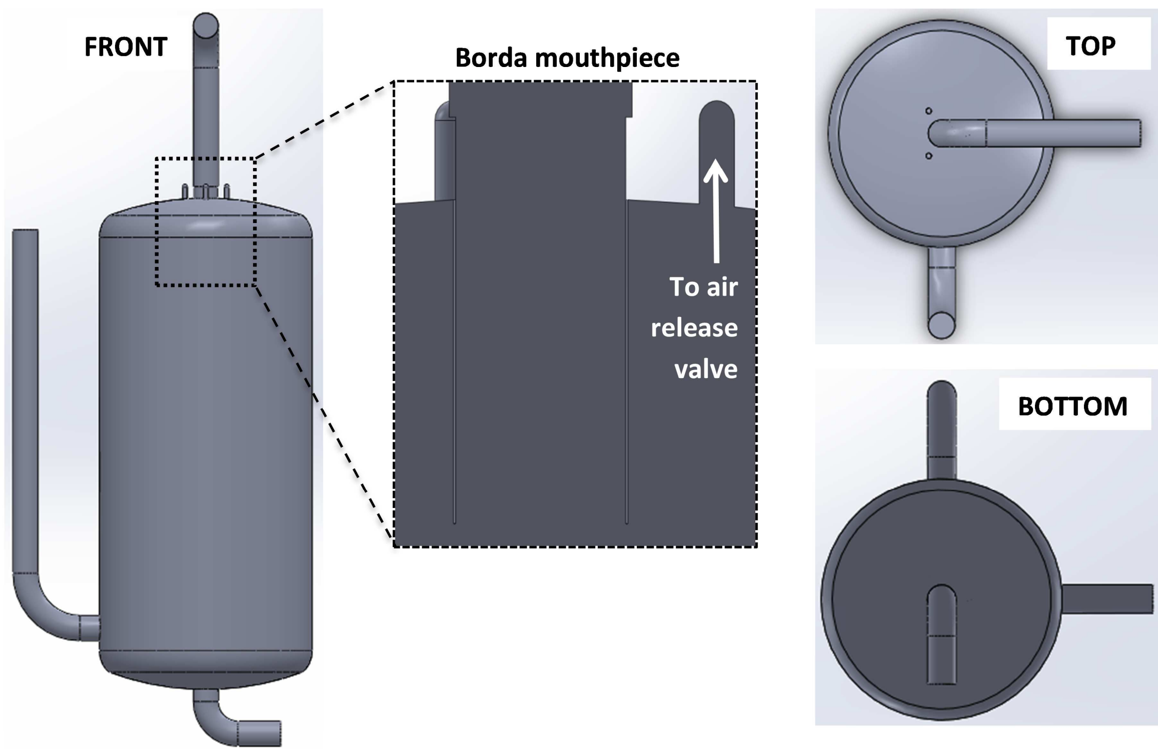

Fig. 10. Front, top and bottom views of the accumulator with the Borda mouthpiece (zoomed section) and the three pipes connecting with the air release valves.

4. 2. CFD model set-up

For the steady state isothermal numerical simulation, a mesh with around 1 million of tetrahedral elements has been used as shown in Figure 11. For modelling the behavior of air bubbles in flowing water, two multiphase models have been used. The Lagrangian Particle Tracking model with the Ishii-Zuber correlation (Ishii et al., 1979) has permitted to obtain complete information on path and residence time of individual bubbles. The Eulerian-Eulerian multiphase model has permitted to calculate the air volume fraction inside the fluid domain. In both cases, a one-way coupling option has been selected which does not take into account the influence of the particles on the continuous liquid phase flow field.

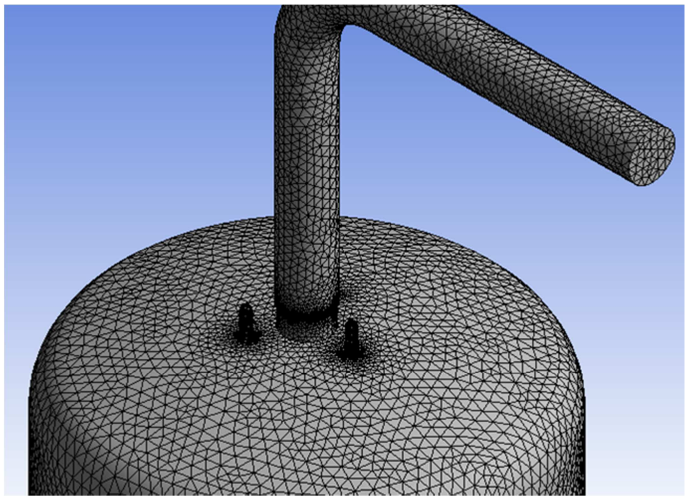

Fig. 11. Detail of the tetrahedral mesh used of the numerical simulation.

To set the boundary conditions at the inlet and two outlet sections, typical operating conditions of the cooling system has been simulated with Flowmaster from which the flow rates and the static pressures have been extracted (see Table 3).

Table 3: Flow rates and total pressures measured at the fluid domain boundaries.

Boundary	***Flow rate (m3/h)***	***Total pressure (bar)***
Inlet	*370.0*	*1.993*
Top outlet	*33.3*	*1.982*
Bottom outlet	*336.7*	*1.978*

The mass flow rate of air particles entering the accumulator through the inlet section has been considered of about 0.411 Kg/s. This figure has been calculated based on a measured oxygen content of about 4000 ppb. A uniform injection has been set with a particle diameter distribution based on considering an equal mass of particles at each diameter from a minimum of 0.1 mm to a maximum of 2.0 mm.

4. 3. CFD results and de-aeration efficiency

The pathlines of randomly injected bubbles in the accumulator for the original and the modified geometry are plotted in Figure 12. As it can be observed on the left of the figure, most of the air particles are exiting the accumulator through the top outlet pipe. On the contrary, the Borda mouthpiece modifies substantially the flow field and the particle trajectories. The entire accumulator becomes a recirculation region and a large number of bubbles are trapped. Moreover it is observed that less particles scape through the top outlet than through the bottom one.

Regarding the effectiveness of the air release valves, the air volume fraction isosurface plotted on the left of Figure 13 clearly indicates that the air bubbles tend to fill the vertical tubes connecting with the valves. Nevertheless, there is not an exact axisymmetric distribution of bubbles on the top of the accumulator and the bubbles tend to concentrate more around one of them. This can be deduced from the air volume fraction values plotted on a cross section plane cutting the tubes as shown on the right of the same figure.

In particular, the de-aeration efficiency can be estimated from the results of the Particle Tracking model. For example, from 100 air particles entered into the modified domain through the inlet pipe only 49 have left at the end of a simulation set with a maximum tracking time of 3500 seconds and a maximum tracking distance of 600 m. Therefore, it can be stated that around a 51 % of air bubbles with diameters between 0.1 and 2 mm are retained in the accumulator. For the particles that exit the domain, around a 2% is through the upper outlet and the rest (96%) is through the bottom outlet.

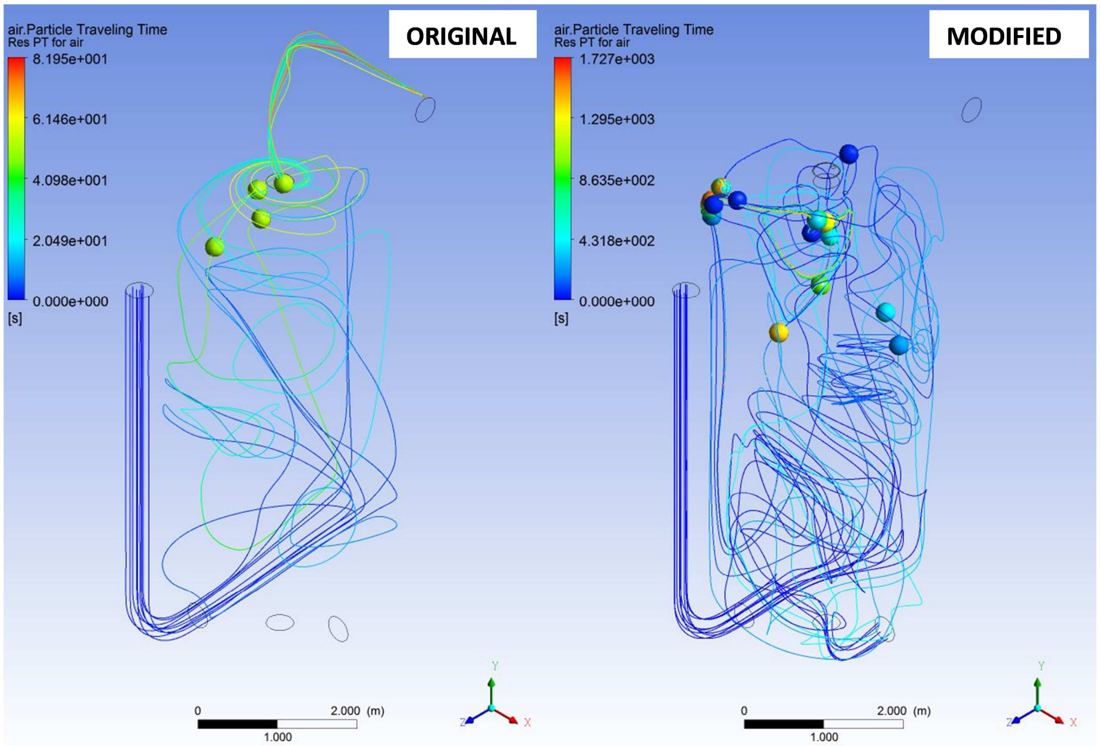

Fig. 12. Bubbles pathlines simulated with the original (left) and the modified (right) domain.

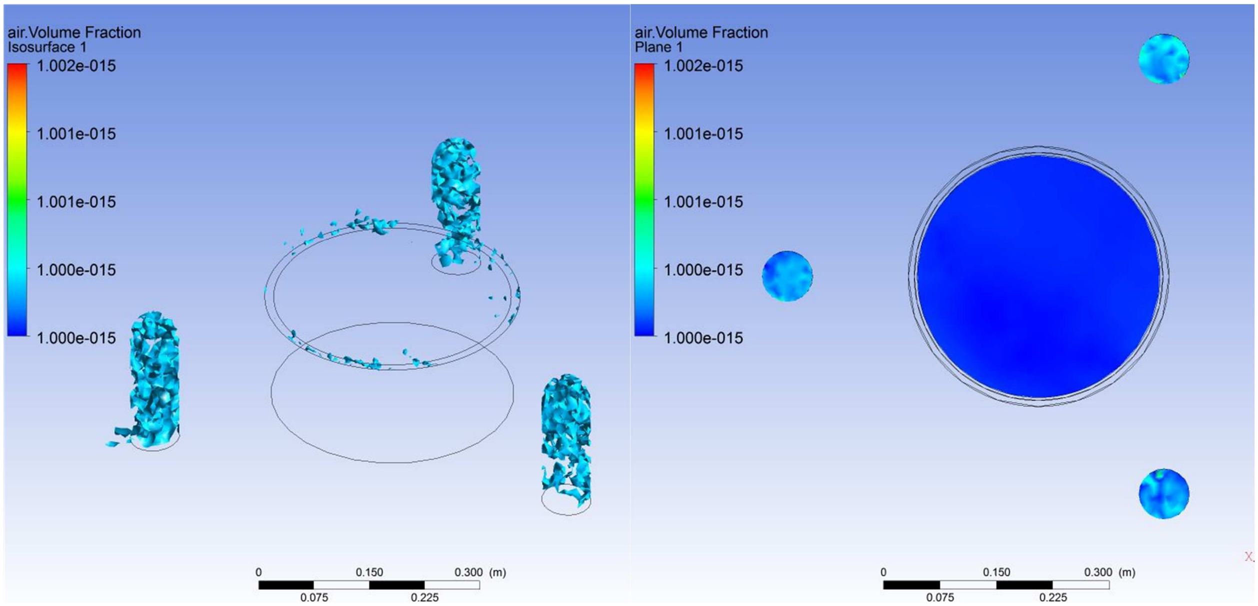

Fig. 13. Isosurface of air volume fraction on the walls of the domain (left) and on a cross section at an intermediate height of the vertical tubes connecting with the air release valves and the outlet top pipe.

5. Conclusions

The thermo-fluid properties of the cooling system have been simulated under typical operation conditions with good accuracy. The model results present maximum deviations from the measured quantities of about 10%.

The global system stability is guaranteed under ideal conditions whenever a positive flow from the accumulator towards the common return exists. Nevertheless, the conditions on the Experimental Area subsystem are significantly affected by changes in Storage Ring or Service Area.

In general, several pipes present mean flow velocities below 0.5 m/s along the entire system. In particular for the Experimental Area consumption ring, a detailed model and simulation has proved a significant velocity rise in most of the pipe sectors when changing the flow distribution from 180° to 360°.

Transient simulations have demonstrated that local changes can affect the whole system behavior due to the common return design configuration.

A numerical simulation with a two-phase model has been carried out to evaluate the effectiveness of a Borda mouthpiece mounted in the upper outlet pipe of the accumulator to promote air collection.

At typical operating conditions, the presence of the modification creates a large recirculation zone that traps more air particles which can be removed with three air release valves. In comparison with the original geometry, around a 50% more of the particles can be retained.

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